A STUDY OF HEAT TRANSFER FROM A ROTATING CIRCULAR CYLINDER

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by

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NOMENCLATURE

A = Area of the cylinder (heat transfer area), ft^2

c_d = Cylinder surface drag coefficient, dimensionless

c, c_p = Specific heat at constant pressure $Btu/lb_m\ {}^\circ F$

d = Diameter of the cylinder

h, h_t = Total heat transfer coefficient, $Btu/hr\,{}^\circ F\,ft^2$

h_c = Pure convection transfer coefficient "

h_r = Radiation heat transfer coefficient "

k, K = thermal conductivity, $Btu/hr\ ft\,{}^\circ F$

q = Heat transfer rate, Btu/hr.

S = $\sqrt{g_c \tau_0/\rho}$ Shear velocity ft/sec.

t = Temperature

v = Velocity of the cylinder ft/hr.

y = distance from the cylinder surface

β = Coefficient of thermal expansion, equal to $1/T$ for a perfect gas

ϵ_M = Eddy diffusivity for momentum ft^2/hr.

ϵ_H = Eddy diffusivity for heat, ft^2/hr.

ν = Kinematic viscosity, μ/ρ ft^2/hr

ρ = Fluid density $lb_m/Cuft$.

τ = Fluid shear stress

μ = Viscosity $lb_m/hr\ ft$.

u^+ = Non dimensional velocity in boundary layer, v/S

y^+ = Non dimensional distance from surface yS/ν

N_{Nu} = $\frac{hd}{k}$, Nusselt's number, dimensionless

N_{st} = $h/v\rho c_p$, Stanton's number "

N_{Re} = Rotating Reynold's number $\frac{dv\rho}{\mu}$ dimensionless

$N_{Pr}(P)$ = Prandtl's number $c_p\ \mu/k$

(G) or N_{Gr} = Greshoff's number $\frac{\beta g \; \Delta t \; d^3 \; \rho^2}{\mu^2}$

U = Maximum free convection velocity at the top of a vertical plate of height d

l = length of a horizontal plate

θ = temperature difference

Subscript

0 = Conditions at Cylinder surface

CONTENTS

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CHAPTER I

INTRODUCTION AND OBJECTIVES

Analysis of the thermal behaviour of the machine elements involves heat conduction through the solid machine parts, radiation between adjacent machine surfaces and then finally heat transfer by convection between solid parts and the surrounding fluid. The problem of heat transfer, when fluids are flowing through or over the stationary surface of machine elements, is not a complicated one since lot of work has been done and much of data are available. However, a rotating surface presents a type of convection problem quite different from convection in ducts or over external stationary surfaces. Such a problem frequently arises in connection with the consideration of thermal behaviour of shaftings, flywheels, turbine rotors and similar machine parts.

Out of the number of geometrical configurations of interest, spherical and cylindrical are two important ones. But even out of these two cylinder perhaps is simpler and industrially more important.

The most elementary of the rotating cylinder problems is the rotating cylinder in an infinite still environment (Air). Regardless of the mechanism in the region very close to the cylinder, heat ultimately is transferred away from th cylinder by free convection. Thus the problem can be termed as the problem of free convection from a rotating cylinder. Many workers viz. Anderson and Saunders (1) , Etemad (2), Dropkin and Carmi (3) have published the results of extensiv

experimental work. All find a strong effect of Grashoff number at low Reynolds number (based on the peripheral velocity of the cylinder) and at very high N_{Re} there is no effect of N_{Gr}. It was found that approximately at Reynolds numbers below 1,000 there was no effect of rotation. Transition occurred at about this N_{Re} followed by a region extending from about 1,000 to 10,000 in which both N_{Gr} and N_{Re} were of importance. Finally above about 10,000 only N_{Re} was of importance.

This is in brief about the previous work on heat transfer from a rotating circular cylinder.

The primary object of the present work has been to check whether the equations given by previous workers for air can be applied to liquids and fluids of higher Prandtl number and if not, to find out a suitable expression for heat transfer from a rotating cylinder to liquids etc.

LITERATURE SURVEY AND PAST WORK

Anderson and Saunders (1) were the first to study this problem in 1953. They took three cylinders of different diameters (1", 1.82" and 3.9") and made some observations with air at 1 atm and 4 atm pressures. The results show that the heat transfer did not increase with speed (i.e. with Reynold's number based on the peripheral velocity) initially but after a critical Reynolds number varied as 2/3rd power of rotational speed. The critical Reynold's number occurred when the peripheral velocity of the cylinder became equal to the velocity of uprising air at the side of the cylinder. This was later confirmed by Dropkin and Carmi (3).

The results were given in the form of plots, log N_{Nu} versus log N_{Re} and it was found that the critical Reynold's number was different for different cylinders (higher for larger diameter cylinders) as expected. The critical Reynold's number at 4 atm pressure was found to be much higher than at 1 atm pressure.

To obtain the approximate value of the velocity of uprising air (for the evaluation of critical N_{Re}) at the side of the cylinder, they used the well known solution for free convection heat transfer from a vertical plate. Assuming the upward velocity approximately same as the velocity at the top of a vertical plate whose height was equal to the diameter of the cylinder, they developed an equation. Taking the maximum free convection velocity at the top of a vertical plate as U, they obtained

$$\frac{U_c \rho d}{k} = .93 \sqrt{GP}$$

or $$(N_{Re})_{crit} = .93 \sqrt{G/P}$$

Since for air the Prandtl Number P has a constant value equal to .73 the above equation became

$$(N_{Re})_{crit} = 1.09 \sqrt{G}$$

Experimental and theoretical results (calculated from the above equation) when compared revealed that theoretical values were slightly lower than the experimental values. The reason for this was the uncertainty of the constant .93 which they said was slightly lower than what it should have been. Since interestingly enough the results were correct qualitatively the equations could be used to predict the heat transfer for the region below the critical Reynold's number.

For the region above the critcal N_{Re} they developed a separate equation by assuming an analogy between the flow around the cylinder and the irregular turbulent flow which occurs in free convection above a heated horizontal plate facing upwards. The centrifugal force normal to the surface in the former case plays an analogous part to buoyancy force in latter case, also normal to the surface. The equation for the heat transfer from a heated horizontal plate facing upwards is

$$\frac{hl}{k} = .14 \left(\frac{ag\theta\, l^3 \rho^2 c}{k\mu}\right)^{1/3}$$

The length l cancells out from both the sides making h the heat transfer coefficient independent of l. To obtain the equation for the heat transfer from a rotating cylinder they replaced the term $ag\theta$, the buoyancy acceleration by v^2/r the centrifugal acceleration. Since by neglecting the thickness of the boundary layer ~~r~~ could be taken as the radius of the cylinder and the circumferential velocity taken at the surface, the expression for a cylinder became

$$\frac{hd}{k} = .14 \left(\frac{2v^2 d^2 \rho^2 c}{k\mu}\right)^{1/3}$$

but $\frac{c\mu}{k}$ for air has a constant value of .73 so the expression became

$$N_{Nu} = .10\ N_{Re}^{\ 2/3}$$

This analogy of course is not correct, because the free convection buoyancy force varies linearly with the excess of temperature of fluid above the surrounding temperature whereas the centrifugal force varies as the square of the tangential velocity relative to the surroundings.

Etemad (2) in 1955 investigated the stability etc. with the help of a Zehndermach Interferometer. The heat transfer data agree well with the experimental results of Anderson and Saunders.

Two cylinders, one of 2 $\frac{3}{8}$ inches diameter copper model

and other 2½ inches O.D. beckelite model were used. Beckelite model was specially used to permit higher rotational speeds and for interferometric study of flow around the cylinder.

Results were given in the form of plots, h versus rpm, N_{Nu} versus (Gr x Pr), N_{Nu} versus N_{Re} etc. The "h versus rpm plot" shows that initially h decreases slightly with speed until it reaches a minimum at a critical speed (70 to 100 rpm) and then increases rapidly at higher speeds. It was found that above 750 rpm there was no effect of temperature difference and the heat transfer coefficient could be nicely predicted by the equation

$$h = .0336\ (rpm)^{.70}$$

N_{Nu} when plotted against (Gr x Pr) with N_{Re} as a parameter gave straight lines with different slopes. The slope varied from .25 at zero rotation to almost zero at a N_{Re} of 8,000, so the line with no rotation could be represented by the equation

$$N_{Nu} = .456\ (Gr \times Pr)^{.25}$$

The "N_{Nu} versus N_{Re} plot" showed that below critical N_{Re} , N_{Nu} varied inversely as N_{Re} and directly as (Gr x Pr) in such a way that the latter was controlling. At a critical N_{Re} of about 800 to 1200 depending upon the magnitude of Gr, N_{Nu} reached a minimum value of about 5% less than the corresponding free convection value. The decrease in N_{Nu} in that range of N_{Re} is due to the laminar couette flow which also increases the effective free convection thermal boundary

At high N_{Re}, free convection effects were negligible and at N_{Re}'s above 8,000 the relationship which was found to hold good was

$$N_{Nu} = .076 \; N_{Re}^{.7}$$

To combine the effects of both natural convection and rotation he plotted N_{Nu} the Nusselt number against $(.5N_{Re}^2 + Gr)Pr$ which gave a straight line with equation

$$N_{Nu} = .11 \; (.5 \; N_{Re}^2 + Gr) \; Pr$$

The stability of flow around the rotating cylinder and more especially the process involved in the transition from laminar couette flow to fully developed secondary flow were investigated by interferometer.

Dropkin and Carmi (3) actually extended the previous work of Anderson and Saunders (1) and Etemad (2). The N_{Re} range they went up to was from 0 to 433,000 whereas the previous workers had only gone up to 65,400.

They took two copper cylinders 3.5 inches O.D. and 4.5 inches O.D. and 2 ft. long with the surface nickel plated to avoid radiation heat transfer. Heating was done by an electric cartridge and temperature measured with the help of cu-constantan thermocouple.

With the help of dimensional analysis they expressed the Nusselt number as a function of different dimensionless groups. The equation obtained was

$$\frac{hd}{k} = c \left(\frac{d^3 \rho^2 g}{\mu^2}\right)^x \left(\frac{v^2}{dg}\right)^y (\Delta\beta)^z \left(\frac{\mu c_p}{k}\right)^m$$

and since for air N_{Pr} is constant so the modified equation for air became

$$\frac{hd}{k} = c_1 \left(\frac{d^3 \rho^2 g}{\mu^2}\right)^x \left(\frac{v^2}{dg}\right)^y (\Delta\beta)^z$$

Various constants and exponents were worked out from experimental results. The "N_{Nu} versus " plot gave straight lines of zero slope indicating no effect of $\Delta\beta$ on N_{Nu}. The "N_{Nu} versus $d^3\rho^2 g/\mu^2$" plot gave straight lines with a constant slope of .35 suggesting an equation

$$N_{Nu} = c_2 \left(\frac{d^3 \rho^2 g}{\mu^2}\right)^{.35}$$

A plot of N_{Nu} versus v^2/dg gave curves for which slope increased and then became a constant (equal to .35). For the region where the slope of the curve was constant they gave an equation

$$\frac{hd}{k} = c_1 \left(\frac{d^3 \rho^2 g}{\mu^2}\right)^{.35} \left(\frac{v^2}{dg}\right)^{.35}$$

and combining the two groups on the R.H.S. obtained

$$N_{Nu} = c_1 (Re)^{.7}$$

"N_{Nu} versus N_{Re} plots" gave curves which indicated that upto a certain value of Reynold's number, the Nusselt number N_{Nu} did not vary with N_{Re}. Then there was transition and after about a N_{Re} of 15,000 there was no effect of N_{Gr}. The results could be represented by

$$N_{Nu} = .073 (Re)^{.7}$$

The overall effect was obtained with a straight line plot

between N_{Nu} and $(.5\ N_{Re}^2 + Gr)$. The equation was

$$N_{Nu} = .095\ (.5\ N_{Re}^2 + Gr)^{.35}$$

Kays and Bjorklund (4) introduced cross flow of air combined with rotation of cylinder and obtained some data. Assuming the entire heat transfer resistance in the rotating boundary layer (applicable only at high rotating velocities where free convection effects and the effects of cross flow are negligible) they developed a heat and momentum transfer analogy.

(i) The analogy solution was quite similar to the Von-Karman analogy as described by Eckert. A laminar sublayer (upto $y^+ = 5$) was assumed followed by a transition region (from $y^+ = 5$ to $y^+ = 20$). After this the diffusivity for heat transfer becomes suddenly very much higher than in the region very close to the surface. Here in this secondary flow is predominent.

(ii) It was assumed that shear stress and heat flux were constants. Velocity distribution in the sublayers was same as that in turbulent boundary layer. Eddy diffusivity of heat was same as the Eddy diffusivity for momentum.

(iii) For $y^+ > 20$ Reynold's analogy was assumed.

Since laminar sublayers are thin shear stress and flux were assumed to be constants. Equations were

$$\tau_o = \frac{\rho}{g_c} (\epsilon_M + \nu) \frac{dV}{dy} \qquad (1)$$

$$(q/A)_o = \rho c_p (\epsilon_H + \alpha) \frac{dt}{dy} \qquad (2)$$

$$\nu = (\epsilon_M + \nu) \frac{du^+}{dy^+} \tag{3}$$

$$Q = (\epsilon_H + \nu/N_{Pr}) \frac{S}{\nu} \frac{dt}{dy^+} \tag{4}$$

They further assumed that $\epsilon_H = \epsilon_M$ so for

Laminar Sublayer: $0 < y^+ < 5$

Assumed velocity distribution was $u^+ = y^+$, then from Eq.(3) $\epsilon_M = \epsilon_H = 0$, so Eq.(4) could be integrated directly for Δt through laminar sublayer. The result was

$$\Delta t_1 = \frac{5Q N_{Pr}}{S} \tag{5}$$

Buffer Layer: $5 < y^+ < 20$

The Von-Karman-Nikuradse velocity profile in the buffer region was assumed

$$u^+ = -3.05 + 5 \ln y^+ \tag{6}$$

Equation (6) was differentiated and substituted in Eq.(3) to give

$$\epsilon_M = \epsilon_H = \frac{\nu y^+}{5} - \nu \tag{7}$$

Eq. (7) was then substituted in Eq.(4) and then integrated from $y^+ = 5$ to $y^+ = 20$ in order to get the expression for the temperature drop

$$Q = \left(\frac{y^+}{5} - 1 + \frac{1}{N_{Pr}}\right) S \frac{dt}{dy^+}$$

$$\Delta t_b = \frac{Q}{S} \int_5^{20} \frac{dy^+}{(y^+/5 - 1 + 1/N_{Pr})} = 5\frac{Q}{S} \ln(3N_{Pr} + 1) \tag{8}$$

Secondary Flow Region: Since eddy diffusivities are much higher than the molecular diffusivities ν and ν/N_{Pr} were neglected in Eqns. (3) and (4). Further it was assumed that the shear stress and heat flux were the same functions of y (rather than that they are the constants as in the sublayers). Eqns. (3) and (4) were not applicable because constant shear stress and heat flux were assumed but their quotient was applicable because of the foregoing assumption. Thus

$$\frac{\nu}{Q} = \frac{\nu}{S}\,\frac{du^+}{dt} \tag{9}$$

If $y^+ = 20$, $u^+ = 12$ from En.(6) thus the temperature drop could be obtained by integrating Eq.(9) from $u^+ = 12$ to $v = v_p$; $u^+ = v_p/S$

$$\Delta t_s = \frac{Q}{S}\int_{u^+ = 12}^{v = v_p;\, u^+ = v_p/S} du^+$$

$$= \frac{Q}{S}\left[\frac{v_p}{S} - 12\right] \tag{10}$$

Total temperature drop thus was

$$\Delta t = \Delta t_1 + \Delta t_b + \Delta t_s$$

$$= \frac{Q}{S}\left[5\,N_{Pr} + 5\ln(3\,N_{Pr} + 1) + \frac{v_p}{S} - 12\right] \tag{11}$$

Expressing this in terms of Stanton number $N_{st}(=N_{Nu}/N_{Re}N_{Pr})$

$$g_c\,\frac{\tau_o}{\rho} = c_p\,(v_p^2/2)$$

$$q/A = h\,\Delta t$$

$$\Delta t = \frac{h\ \Delta t}{c_p v_p \rho \sqrt{c_d/2}} \left[5\ N_{Pr} + 5\ \ln\ (3N_{Pr} + 1) + \frac{v_p}{v_p\ \sqrt{c_d/2}} - 12 \right]$$

$$N_{st} = \frac{\sqrt{c_d/2}}{\left[5N_{Pr} + 5\ \ln\ (3\ N_{Pr} + 1) + \frac{1}{\sqrt{c_d/2}} - 12 \right]} \qquad (12)$$

$$N_{Nu} = \frac{N_{Re}\ N_{Pr}\ \sqrt{c_d/2}}{\left[5N_{Pr} + 5\ \ln\ (3N_{Pr} + 1) + \frac{1}{\sqrt{c_d/2}} - 12 \right]} \qquad (13)$$

Use of Eqn.(12) and (13) required knowledge of c_d, the friction coefficient. These could be obtained by the experimental results of Theodorsen and Regier (7). The results can be expressed as

for $N_{Re} \sqrt{c_d} > 950$

$$\frac{1}{\sqrt{c_d}} = -1.828 + 1.77\ \ln\ N_{Re} \sqrt{c_d} \qquad (14)$$

for $N_{Re} \sqrt{c_d} < 950$

$$\frac{1}{\sqrt{c_d}} = -\ 3.68 + 2.04\ \ln\ N_{Re} \sqrt{c_d} \qquad (15)$$

Eqn. (13) together with (14) corresponded closely to the empirical equation proposed by Dropkin and Carmi.

Since the boundary layer was quite thick at low N_{Re}, they traced the temperature profiles by using a 30 gauge Iron-Constantan thermocouple wire, and compared them with theoretical profiles. It was found that profiles in laminar sublayer and buffer region matched very nicely but in secondary flow region there was slight difference. The actual profile was slightly steeper than the experimental one.

Merely by intution they extended the formula proposed

by Etemad (2) and proposed an equation for cross flow combined with rotation. The equation was

$$N_{Nu} = .135 \left[(.5\ N_{Re}^2 + N_{Rs}^2 + N_{Gr})\quad N_{Pr} \right]^{1/3} \tag{16}$$

for N_{Gr} and N_{Rs} negligible as compared to N_{Re} and for $N_{Pr} = .7$ (Air); Eqn. (16) reduced to

$$N_{Nu} = .095\ N_{Re}^{2/3} \tag{17}$$

This Eqn.(17) closely corresponded to equation proposed by Anderson and Saunders (1).

PRESENT WORK

The previous workers (Anderson and Saunders (1), Etemad (2) and Keys (4) etc.) have done work with air only and no work whatsoever with fluids other than air has been reported. Air firstly is the simplest fluid to work with and does not offer many experimental difficulties which other fluids do offer. Secondly the availability of air in unlimited quantity is really an asset for experimentation. Of course data with air is of utmost importance, still the work with fluids other than air cannot be in anyway underestimated.

The aim of the present work has been to see whether the equations given for air could be used satisfactorily for liquids as well and if not, to find, a suitable expression.

There are few points which can come in way to the general use of the equations proposed for air. These points are

(i) Some of the assumptions the previous workers made are not correct for liquids. The boundary layer for example cannot be assumed to be thin even at very high velocities of rotation in fluids of high Prandtl number as in Kays (4) expression. This is because, the convection transfer is quite significant in high viscosity fluids and the boundary layer is thicker.

(ii) The expression given by Anderson and Saunders (1) were based on certain analogies which were not exact.

(iii) Whole of the previous work is based on the experiments with air.

The above points do not in anyway permit the general application of the expressions given by previous workers so a test was necessary. With this aim the present work has been carried out.

CHAPTER II

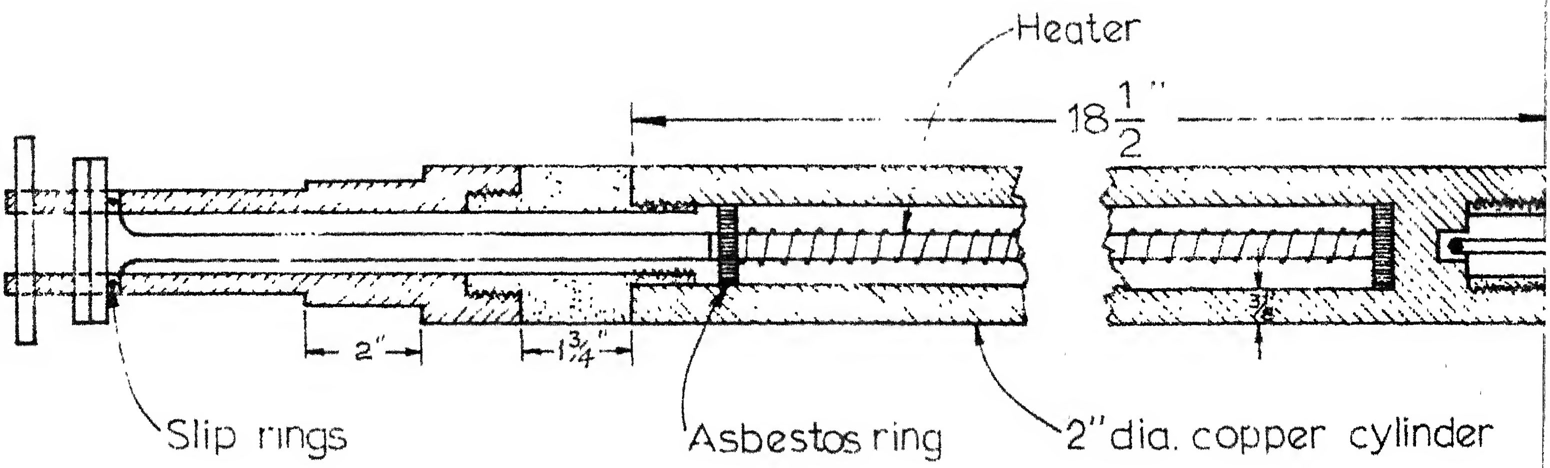
EXPERIMENTAL SETUP

A copper cylinder of 2 inches O.D. and approximately 19 inches length was taken as an active test section. It was heated by nichrome wire (wound on a fireclay rod) electric radiation heater of 1000 watts power which was supported on two asbestos rings which nicely fitted inside the copper cylinder (as shown in the figure). The copper cylinder as a matter of fact was not hollow throughout but some of its portion was solid in between which separated the heater section and the temperature measuring section. The remainder of the cylinder consisted of two teflon spacers (which served two purposes (i) as thermal insulators (ii) as indirect support for the heating rod) and brass shafts supported on two ball bearings placed in two bearing housings.

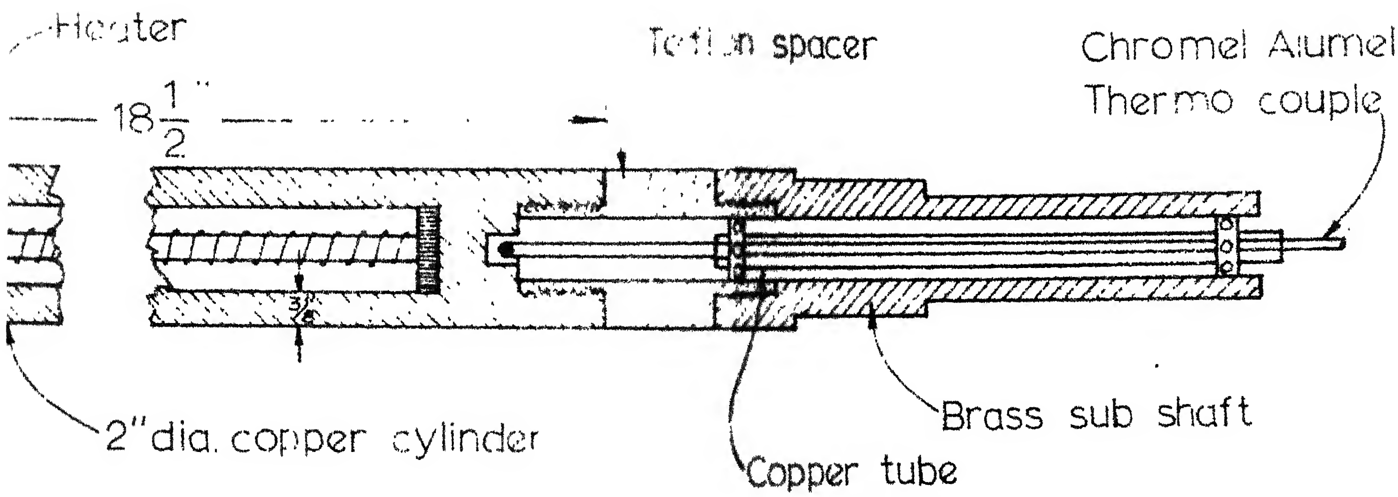
The whole shaft was put in a round bottom mild steel tank by attaching the bearing housing to the tank walls. This tank was jacketed by a water jacket with 1 1/8" inlet at the bottom and two outlets at the top for water. This was done to make the whole system more ideal and to increase the outer heat transfer coefficient for rapid attainment of steady state.

In operation, A-C power was supplied to the heater circuit through brass brushes and slip rings from an Auto-transformer controlled 230 volts source. The input power was measured by an A-C watt meter of 0 to 1500 watts range.

TEST CYLINDER



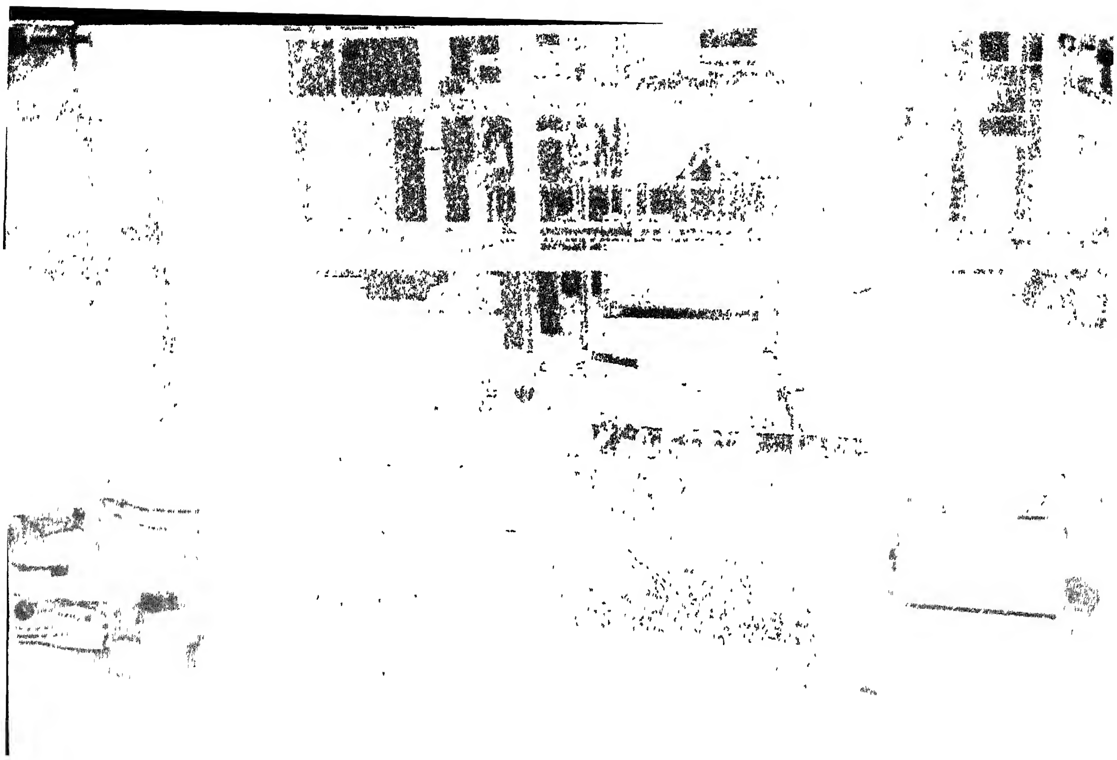
TEST CYLINDER



The cylinder was rotated by connecting it to an A.C./D.C. motor whose speed was controlled by another autotransformer, thereby controlling the speed of cylinder. Higher speeds (upto only 700 rpm) were obtained by directly coupling the motor with the cylinder but lower speeds could not be obtained by direct coupling because of the higher friction in the system. The lower speeds were obtained by a worm-gear arrangement; worm being rotated by motor and gear connected to the cylinder. A mechanical tachometer was used to measure the speed of the cylinder and since the diameters of the tachometers' disc and that of brass subshaft, were same it directly gave the rpm.

The temperature of the cylinder's surface was sensed by a stationary thermocouple which projected into a cavity in the rotating copper test section as shown. To minimize the temperature difference between the copper cylinder and the thermocouple junction, the tip was flattened to provide some stirring effect for increased convection heat transfer. This thermocouple (Chromel-Alumel) passed through a small copper tube which itself was supported by two ball bearings. To make sure that the thermocouple was in position, two rubber stoppers with holes drilled in were put. Thus the whole cylinder was rotating while the thermocouple was stationary.

It was tested whether the internal thermocouple read the same temperature as the external surface temperature by soldering two thermocouples lengthwise along the surface of the cylinder and comparing their readings with the readings by internal thermocouple. It was found that the internal



thermocouple read essentially the same temperature as read by the thermocouples at the surface of the cylinder. In order to minimize the axial temperature drop the copper cylinder was made quite thick (wall thickness was about 1/2") it was tested also. All the temperatures (i.e. the temperature of the cylinder's surface and temperatures of water along the tank wall) were measured by a multi purpose Honeywell Potentiometer.

Water leakage through the bearings initially was a big problem and many devices had to be tried. Finally it was prevented by providing two spring loaded (tension of the spring could be adjusted) rubber O-rings which pressed hard against the internal surfaces of the bearing housings.

CALIBRATION OF THERMOCOUPLES

All the thermocouples were calibrated with the help of a thermostat (0 to 100°C range). The thermocouples were dipped in a mercury contained glass test-tube which itself was dipped in water. An accurate (accuracy upto .1°C) thermometer was also dipped in the test-tube to give the temperature of the mercury therein. All the thermocouples were found to read the temperature correctly. Since measurements of temperature difference and not absolute values of temperature were required, this calibration method was sufficiently accurate.

CHAPTER III

MEASUREMENTS WITH AIR

To test the thermal behaviour of the apparatus runs were taken in air and compared with reported data.

Natural Convection (Cylinder Non-Rotating):

The free convection test results are presented graphically in Figure No.1 as plots of Nusselt number versus the product of Grashoff and Prandtl numbers. Included for comparison are the recommended equations of McAdams (5) based on a large amount of data from various sources and the data obtained by Etemad (2) from his cylinder when not rotating. On an average the data obtained very nicely match with the McAdams curve. The experimental uncertainty expected in the stationary tests is somewhat greater than expected for tests with rotation, but of probably more significance is the difficulty of attaining steady state because of the large time constant (approximately 4 hours). It is this fact that causes the scatter of test points.

In order to provide a check on the adequacy of the experimental apparatus runs were taken with cylinder outside the tank as well as cylinder inside the tank and compared. It was found that cylinder in tank gave essentially the same results as outside the tank.

All the properties of air were calculated at a temperature half way between (the arithmatic mean) the cylinders surface temperature and ambient temperature. It was found that approximately 40% of the total transfer took place by radiation heat transfer. The total heat transfer coefficient values were quite

low (about 2 to 3 Btu/hr ft^2°F) with temperature differences ranging from 50°F to 150°F. The observations taken are tabulated along with the calculated values of Nusselt, Prandtl, and Grashoff numbers in table no.1.

Table 1

Sl. No.	Watts Power	Temp. Cylinder mV	°C	Ambient Temp °C	Δt °F	N_{Nu}	N_{Pr}	N_{Gr}	$N_{Gr}N_{Pr} \times 10$
1	38	2.832	69.5	31	69.3	13.16	.683	5.02×10^5	3.435×10^6
2	40	2.58	63.5	31	58.5	18.45	.684	4.36×10^5	2.986×10^6
3	48	3.145	77.0	31	82.8	12.9	.68	5.7×10^5	3.880×10^6
4	100	4.57	111.5	30	146.7	15.85	.682	9.380×10^5	6.40×10^6

The value of ϵ_T the total emmissivity was taken for copper with a thick oxide coating from Perry's Hand Book (8).

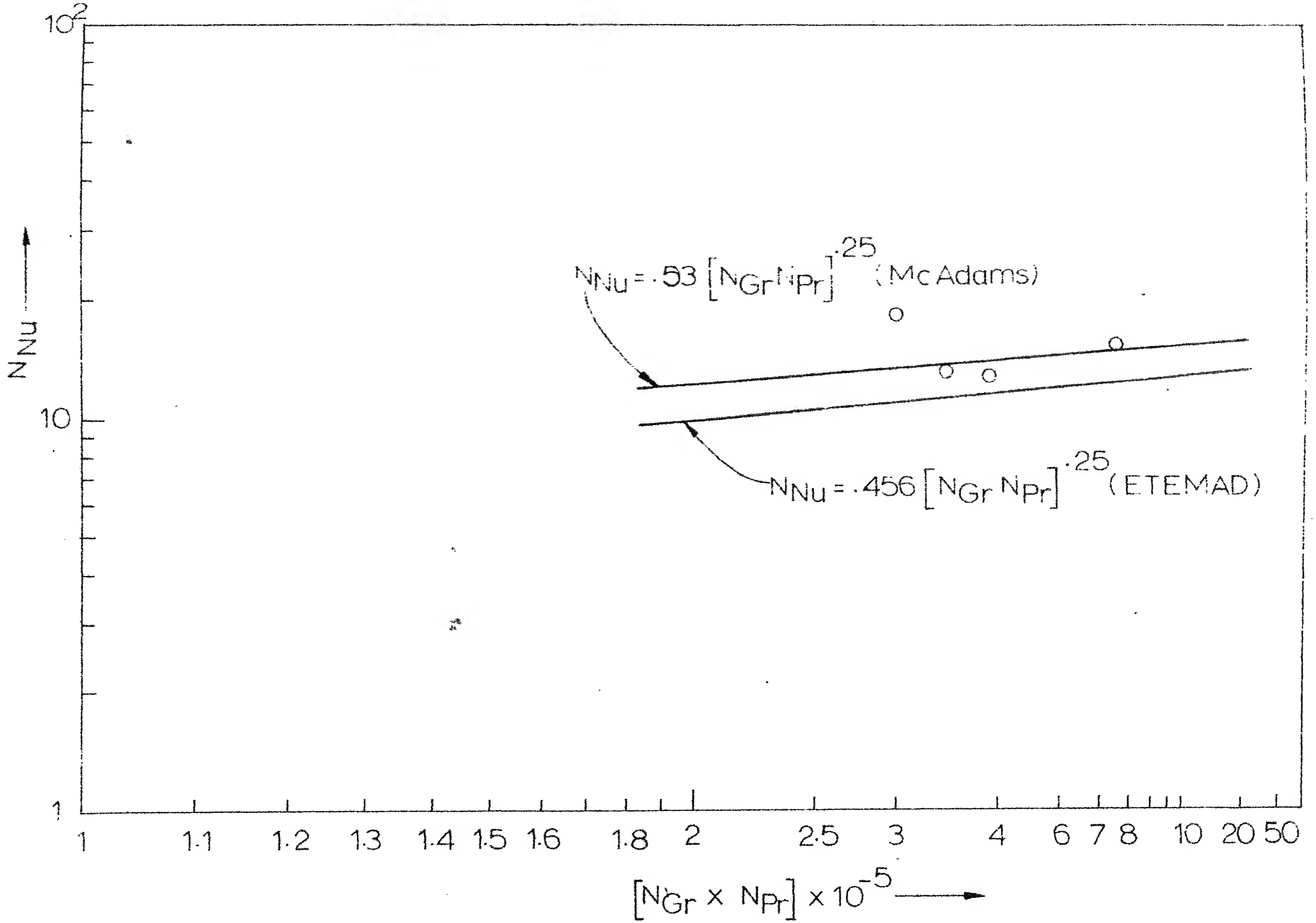


Fig. 1.

Natural convection heat transfer to air from a cylinder

Forced Convection (Cylinder Rotating):

Several observations were made with the cylinder rotating in air inside the tank. The cylinder speed was varied from about 60 rpm to 620 rpm (higher speeds could not be obtained because the motor being used had a maximum speed of 700 rpm) correspondingly varying the Reynold's number from 449 to 5000. All the fluid properties were calculated at a temperature half way between the cylinder's surface temperature and the ambient temperature. The test results are presented graphically in the figure no.2 where Nusselt number is plotted as a function of Reynold's number. The results quite nicely match with the results obtained by Kays and Bjourklund (4).

At lower speeds as shown there is a slight difference between these results and the results obtained by Kays and Bjourklund (4) but at higher speeds of rotation, the two results tend to match nicely.

The tabulated results are given in table no.2.

In this case also about 25 to 30% of the total heat transfer took place by radiation. The time constant for steady state attainment in this case also was about 3 hours.

Table No. 2

Sl. No.	Power (Watts)	Temp. Cylinder mV	°C	Ambient temp. °C	△ t °F	Speed rpm	N_{Nu}	N_{Re}
1	23	2.2	55	33	39.6	329	14.67	2630
2	23.5	2.125	52.5	35	31.5	495	22.85	3955
3	34	2.58	63.5	35.5	50.4	452	18.51	3510
4	37	2.36	58.25	36.5	39.15	620	30.8	4744
5	47	3.19	78.25	34	79.65	60	13.92	449
6	51	3.132	76.8	34	99	276	16.89	1098
7	70	3.43	84	34	138.2	405	21.48	3000
8	75	3.636	89	34	99	235	16.78	1728
9	97	4.195	102.2	34	122.8	145	20.47	1032

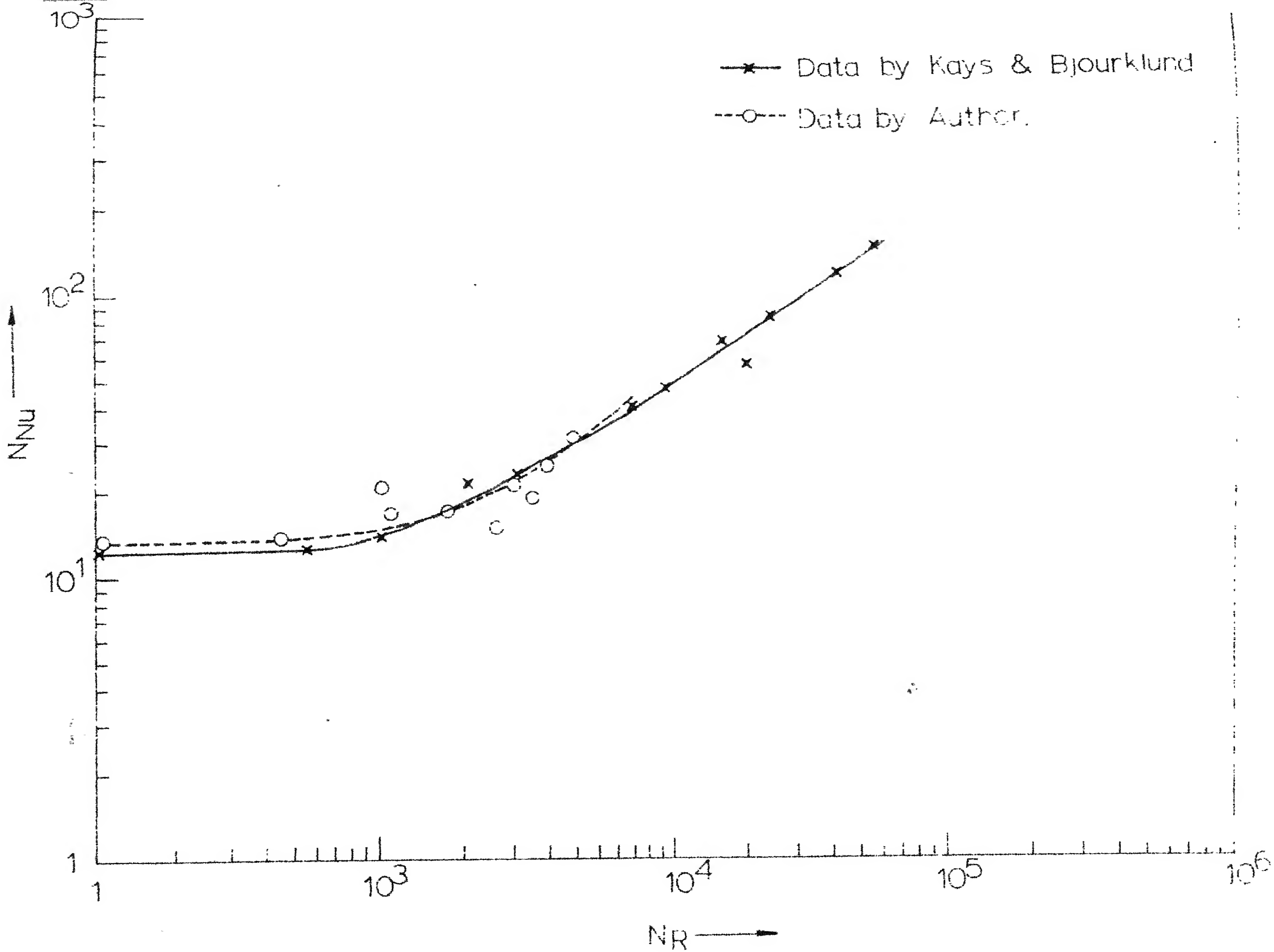


Fig.2.
Heat transfer from a rotating cylinder to air

CHAPTER IV

OBSERVATION WITH WATER

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It was found that the temperature near the water surface was quite different than the temperature at the bottom of the tank. The temperature of water near the water surface and temperature at the side of the tank wall were quite higher than the temperature at the bottom which was almost atmospheric all the time during natural convection heat transfer.

Time for the steady state attainment in case of water was also high (approximately 3 hours).

The results matched nicely with result of Ackerman(5).

Cylinder Rotating: Temperatures around the cylinder in water were quite different in rotating cylinder case also specially at moderate speeds, so five thermocouples were placed along the inside of the tank wall to measure the different temperatures. As expected the temperatures at the bottom were quite low than the temperatures at the top (this is of course because of the natural convection heat transfer which causes the heated fluid to go up). The thermocouples were put only at the inside of the tank wall and no thermocouple was put right above the cylinder in water because of the fact that temperature averaging was to be done for the temperatures along the tank wall. This is because almost whole of the heat transfer (or heat loss)

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takes place through the tank walls because of much higher heat transfer coefficient. The heat transfer to the surrounding air through the open exposed water surface is much lower because of the lower heat transfer coefficient.

In order to make the system more ideal and to facilitate rapid heat transfer (for rapid steady state attainment) water was circulated in the jacket surrounding the tank. The flow rate of water in the jacket was kept quite high and because of this even at maximum power input, there was not appreciable increase in the water temperature. The idea was that if wall and water film on the outside of tank wall did not offer much resistance to heat transfer, the average temperature of water jacket could be used as the mean average temperature of water with which the temperature difference was to be calculated. Unfortunately this did not happen and to find an average temperature of water, five thermocouples had to be placed along the wall. These thermocouples gave different readings so to find an average water temperature, temperature versus angle (θ) plot was made and then with the help of graphical integration an average temperature was found out. This mean average temperature was called "theta mean average temperature $t_{\theta_{avg}}$". The thermocouple configuration is shown in figure no.4. The position of one thermocouple (in the figure, position of no.1) was taken as to be $\theta=0$ and then corresponding to that thermocouple positions of other thermocouples were found out. The angles were $\theta_{12}, \theta_{13}, \theta_{14}, \theta_{15}$ etc. The values of different angles in actual experiment were

$\theta_{11} = 0$ $\theta_{12} = .1772\ \pi$

$\theta_{13} = .4\ \pi$ $\theta_{14} = .677\ \pi$ $\theta_{15} = 1.032\ \pi$

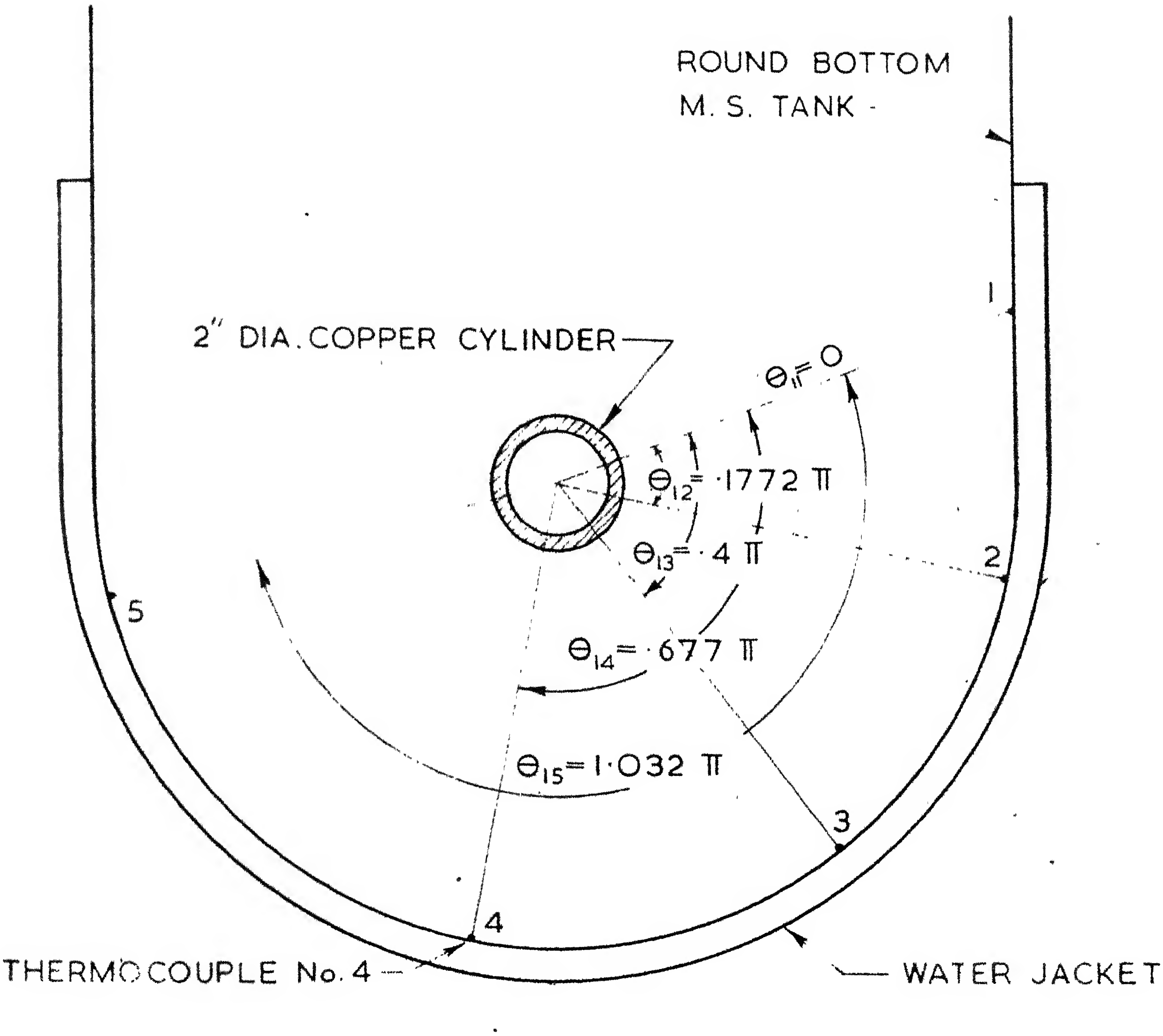
Seven or eight runs were taken from zero rpm to about 485 rpm which correspond to Reynold's number values of 0 to 8.25×10^4. It was found that after about 200 rpm, the flow becomes quite turbulent and the temperature of water throughout the tank tends to equalize (i.e. all the five thermocouples read, more or less the same temperature). So for higher Reynold's number values the arithmatic average temperature and the "theta mean average temperature" were same.

Average temperature of water at which the properties of water were to be evaluated was obtained by stirring the water immediately after taking observations. During that period power was shut down and the rotation stopped.

The two plain faces of the tank were insulated with magnesia in order to minimize the heat loss through those plain faces.

All the observations are tabulated with different temperatures read by five thermocouples along with the calculated "theta mean temperature $t_{\theta_{avg}}$" obtained after graphical integration,(Tables No.3 and 4)

Results are plotted on the graph as plots of Nusselt number (N_{Nu}) versus $(.5N_{Re}^2 + Gr)Pr$ as shown. The plot is a straight line which matches with the line obtained by Kays Bjorklund (4) and Etemad (2) with air.



POSITION OF THERMOCOUPLES

FIG No 4

Thus the equation which was proposed by Etemad (2) for air holds good for water also and probably could be used to predict the heat transfer coefficients in a rotating cylinder case for fluids of higher Prandtl number

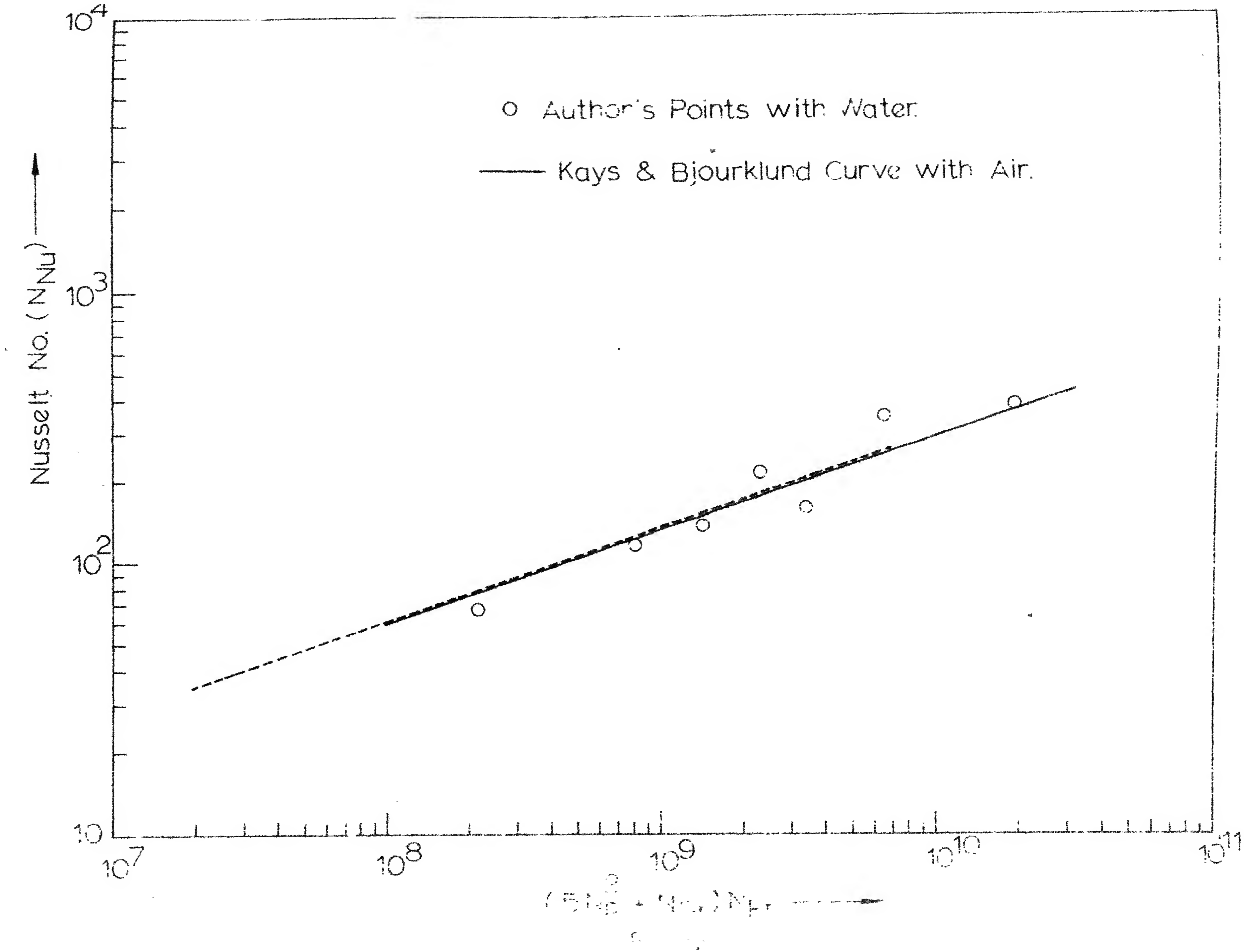
TABLE 3

Flow rate of the jacket water = .474 liters/sec.

Sl. No.	Power (Watts)	Temp. Cylinder mV	°C	Speed (rpm)	Temperatures at the tank wall t_1	t_2	t_3	t_4	t_5	Avg. temp. of water after stirring
1	54x4	1.657	41.4	0	38	35.5	35	34.9	37.1	
2	"	1.525	38	200	36.7	36.7	36.7	36.15	36.15	37°C
3	212x4	2.08	51.5	0	46.4	37	35.25	35	41.5	
4	"	1.78	44.25	123	43	42	35.3	35	39.5	41°C
5	91x4	1.595	39.5	0	35.5	31	30.25	30	32.5	
6	"	1.38	34.5	95	33.5	33	30.75	30	32.75	32.5°C
7	146x4	1.46	36.5	165	35	35	34	32.5	32.75	35°C
8	"	1.424	35.6	293	34.1	34.1	34	33.4	33.5	34.25°C
9	147x4	1.42	35.5	485	34.1	34.1	34	33.75	33.75	34.25°C

TABLE 4

Sl. No.	Power (Watts)	Speed (rpm)	t_{O} avg °C	N_{Nu}	N_{Pr}	$N_{Re}x10^4$	N_{Gr}	$(.5N_{Re}^2+Gr)Pr$
1	54x4	200	36.5	155.1	4.97	3.6	$.1094x10^8$	$3.272x10^9$
2	212x4	123	37.5	134.8	4.57	2.41	$.576x10^8$	$1.59\ x10^9$
3	91x4	95	31.2	119.7	5.46	1.567	$.208\ x10^8$	$7.825x10^8$
4	146x4	165	33.6	217.5	5.18	2.86	$.1971x10^8$	$2.22x10^9$
5	146x4	293	33.8	371	5.28	4.985	$.1125x10^8$	$6.64x10^9$
6	147x4	485	33.94	385	5.28	8.25	$.965x10^7$	$1.802x10^{10}$



Author's Points with Water.
Kays & Bjourklund Curve with Air.
Nusselt No. (N_{Nu})
10^4
10^3
10^2
10
10^7
10^8
10^9
10^{10}
10^{11}

CHAPTER V

CONCLUSIONS AND RECOMMENDATIONS

A. Conclusions:

Following conclusions are drawn from the present investigation:

i) The equations given by previous workers for the prediction of heat transfer in rotating cylinder case for air can be satisfactorily used for liquids of moderate Prandtl numbers.

ii) About 30 to 40% of the total heat transfer takes place by radiation in case of air while working with unpolished copper surface.

iii) This type of averaging (the avereging with the angle θ) is a satisfactory method for finding out an average temperature of liquid for the temperature difference determination in this kind of experiment.

B. Recommendations:

i) The equation can only be generalized when it proves alright for very high viscosity fluids such as mobil oils etc., so some observations with mobile oils must be taken.

ii) The apparatus with slight modification can be used to study the effect of rotation on boiling heat transfer coefficients. For this steam must be used as a heating medium

APPENDIX 'A'

SAMPLE CALCULATION FOR AIR

1. Cylinder Not Rotating (Natural Convection)

With observation no.2

Power input = 40 watts = 40x3.408 Btu/hr.

Temperature of the cylinder t_s = 63.5°C = 146.3°F

Temperature of air t_a = 31°C = 87.8°F

Average temperature $= \frac{146.3 + 87.8}{2} = 117.05°F$

Temperature difference $\Delta t = 146.3 - 87.8 = 58.5°F$

Area of heat transfer $= \frac{\pi \times 2 \times 37}{12 \times 24} = .806 \text{ ft.}^2$

The properties of air are

Thermal conductivity K (9) = .0168 Btu/ft.°F hr.

Specific heat c_p(10) = .25 Btu/lb.°F

Viscosity μ (11) = .04598 lb_m/ft.hr.

$$\text{Prandtl number} = \frac{\mu c_p}{k} = \frac{.04598 \times .25}{.168} = .6845$$

Assuming the air to be ideal gas the density would be

$$= \frac{29 \times 492}{359 \times (460+117.05)} = \frac{29 \times 492}{359 \times 577.05} = .0688 \ lb_m/\text{Cuft}$$

Grashoff's number $N_{Gr} = \frac{\beta \Delta t g d^3 \rho^2}{\mu^2}$

$$= \frac{58.5 \times 32 \times 600 \times 100 \ (.0688)^2}{(577.05) \times (.046)^2}$$

$$= 4.36 \times 10^5$$

The equation for heat transfer is

$$q_t/A = h_c \Delta t + .1714 \ \epsilon_T \left[\left(\frac{T_s}{100}\right)^4 - \left(\frac{T_a}{100}\right)^4 \right]$$

putting different values

$$\frac{40 \times 3.408}{.806} = h_c \times 58.5 + .1714 \times .78 \left[\left(\frac{460 + 146.3}{100}\right)^4 - \left(\frac{460+87.8}{100}\right)^4 \right]$$

or

$$h_c = 2.89 - \frac{.1714 \times .78}{58.5} \left(6.063^4 - 5.478^4 \right)$$

$$= 2.89 - \frac{.1714 \times .78}{58.5} (1350 - 900)$$

$$= 2.89 - 1.029 = 1.861$$

Nusselt number $N_{Nu} = \frac{h_c d}{k} = \frac{1.861 \times 1}{6 \times .0168} = 18.45$

$(N_{Gr} \times N_{Pr}) \times 10^{-5} = 2.986$

2. Cylinder Rotating:

With observation no.9

Total power input = 97 watts = 97 x 3.408 Btu/hr.

Temperature of cylinder t_s = 102.25 °C = 216°F

Temperature of air t_a = 34°C = 93.2°F

Temperature difference Δt = 216 - 93.2 = 122.8°F

Area of heat transfer A = .806 ft^2

Total heat transfer coefficient $h_t = \frac{97 \times 3.408}{122.8 \times .806}$

$= 3.34$ Btu/hr ft^2°F

Average temperature $= \frac{216 + 93.2}{2} = 154.6$°F

The properties of the air are

Specific heat c_p = .2375 Btu/lb_m°F

Thermal conductivity k = .01698 Btu/hr ft °F

Viscosity μ = .0196 centipoises = .04745 lb_m/ft hr.

Density of air (assuming air to be an ideal gas)

$$= .06451\ lb_m/Cuft$$

Prandtl number $N_{Pr} = \frac{c_p \mu}{k} = \frac{.2375 \times .04745}{.01698} = .664$

Velocity of rotation $v = \frac{\pi \times 145}{6 \times 60}$ ft/sec.

$$= 1.263 \text{ ft./sec}$$

Reynold's number $N_{Re} = \frac{dv\rho}{\mu} = \frac{1x1.263x3600x.06451}{6x.04745}$

$$= 1032$$

From the equations given previously we get

$$h_c = 3.34 - \frac{.1714 \times .78}{122.8}\left[\left(\frac{460 + 216}{100}\right)^4 - \left(\frac{460 + 93.2}{100}\right)^4\right]$$

$$= 3.34 - \frac{.1714 \times .78}{122.8}\left(6.76^4 - 936\right)$$

$$= 3.34 - \frac{.1714 \times .78}{122.8} \times 1150$$

$$= 3.34 - 1.253 = 2.087 \text{ Btu/hr ft}^2\,{}^\circ F$$

Nusselt number $N_{Nu} = \frac{h_c d}{k} = \frac{2.087 \times 1}{6 \times .01698}$

$$= 20.47$$

APPENDIX 'B'

SAMPLE CALCULATION FOR WATER

Cylinder Rotating   With observation no.2

Power input = 212 x 4 watts = 212x4x3.408 Btu/hr.

Speed of revolution = 123 rpm

Area of heat transfer = .806 ft^2

Temperature of the cylinder = 44.25°C

Theta mean average temperature of water = 37.5°C

Temperature difference $\Delta t = 44.25 - 37.5 = 6.75$ °C = 12.15°F

Total heat transfer coefficient $h_t = \frac{q}{A\ \Delta t}$

$$= \frac{212 \times 4 \times 3.408}{.806 \times 12.15} \text{ Btu/hr ft}^2 \text{°F}$$

$$= 295$$

Average temperature of water obtained after stirring

= 41°C = 105.8°F

The properties of water at this temperature are

Specific heat c_p = 1Btu/lb_m°F ; Viscosity = .688 x 2.42 Lb_m/ft h

Thermal conductivity k = .3645 Btu/hr ft°F

Nusselt number $N_{Nu} = \frac{hd}{k} = \frac{295 \times 1}{6 \times .3645} = 134.8$

Prandtl number $N_{Pr} = \frac{c_p \mu}{k} = \frac{1 \times 2.42 \times .688}{.3645} = 457$

Grashoff number $N_{Gr} = \frac{\beta \Delta t\, g d^3 \rho^2}{\mu^2}$

$$= \frac{1 \times 12.15 \times 32 \times 3600 \times 3600 \times 1 \times 62.5^2}{(460 + 105.8) \times 6 \times 6 \times 6 \times (1.665)^2}$$

$$= \frac{12.15 \times 32 \times 600 \times 100 \times (6.25)^2 \times 100}{565.8 \times (1.665)^2}$$

$$= \frac{12.88 \times 32 \times (6.25)^2 \times 10^4}{(1.665)^2}$$

$$= .576 \times 10^8$$

Velocity of rotation $v = \frac{\pi \times 1 \times 123}{6 \times 60}$ ft/sec.

Reynold's number $N_{Re} = \frac{dv\rho}{\mu}$

$$= \frac{\pi \times 123 \times 3600 \times 625}{6 \times 6 \times 60 \times 1.665}$$

$$= \frac{3.14 \times 1.23 \times 1.04 \times 10^4}{1.665}$$

$$= 2.41 \times 10^4$$

$$.5N_{Re}^2 = .5(2.41 \times 10^4)^2 = 2.905 \times 10^8$$

$$(.5N_{Re}^2 + Gr)Pr = (2.905 + .576) \times 4.57 \times 10^8$$

$$= 1.59 \times 10^9$$

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(11) John Perry "Chem.Engg. Handbook" p.3-197, Fig.3-42.